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Lubrication

A Technical Publication Devoted to the Selection and Use of Lubricants

THIS ISSUE

INDUSTRIAL
BEARING
LUBRICATION

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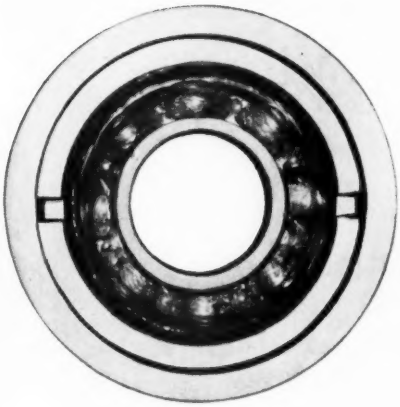

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LUBRICATION

A TECHNICAL PUBLICATION DEVOTED TO THE SELECTION AND USE OF LUBRICANTS

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INDUSTRIAL BEARING LUBRICATION

THE principle of using a lubricant to reduce friction between moving parts dates back to antiquity. Archaeologists unearthed an ancient chariot from about 1400 B.C. and found that the wheels had been lubricated with beef or mutton tallow. Even today such fats continue to be used on ox carts in some backwoods areas. But whether it be wagon wheels turning at a few RPM or jet engine compressors spinning at 50,000 RPM, the principle by which a lubricating film functions is the same; a viscous layer separates the moving parts. This principle now has been amplified only in the degree in which care is taken to insure that the best available lubricant is used for the application in hand.

BEARING DESIGN

With a great variety of lubricants now available and with a well-established theoretical background in bearing performance, bearings now are designed and lubricated in a rational manner rather than by empirical rules of thumb. However, a complete and accurate mathematical solution for the performance of a journal bearing is very complex. Many factors must be considered; some of them are discussed in this issue. The general theoretical bearing analysis must include:

1. Load, fluctuations and directions.
2. Bearing length, width and shaft clearance.
3. Oil grooving and location of oil inlet hole.
4. Side leakage.
5. Temperature of the oil inlet, of the bearing support and journal.
6. Oil characteristics, particularly its viscosity at the operating temperature and pressure.

Mathematical treatments are covered by many texts[1] on the subject, accordingly none will be presented here. By such an analysis, however, one can predict the performance of an industrial bearing. In fact, when all limits are known for a large mill bearing, it is quite practical for the viscosity of a lubricant to be specified on the basis of such mathematical calculations.

Special test machines frequently are used by bearing designers to investigate basic bearing operations. Such a typical machine is shown in Figure 1. This tester can investigate a wide variety of bearing configurations, effects of bearing clearance, oil grooving, etc., both with oil and grease lubrication. Results of a typical experimental run in a bearing tester are shown in Figure 2, which presents the pressure distribution of oil within the bearing clearance. The bearing arrangement can be represented diagrammatically as supporting a high-speed rotating shaft of weight "W." The oil supply hole is at the top and the point-to-point pressure distri-

[1]Mathematical analyses of bearing performance are covered by: Shaw, M. C. and Macks, E. F., "Analysis and Lubrication of Bearings," McGraw Hill 1949; Norton, A. E., Lubrication, McGraw Hill, 1942; Muskat, M., and F. Morgan: "The Theory of the Thick Film Lubrication of a Complete Journal Bearing of Finite Length," Jour. Applied Physics, 9, 393 (1938); Christopherson, D. G., "A New Mathematical Method for the Solution of Film Lubrication Problems," Proc. Inst. Mech. Engrs. (London), v. 146, 126 (1942).

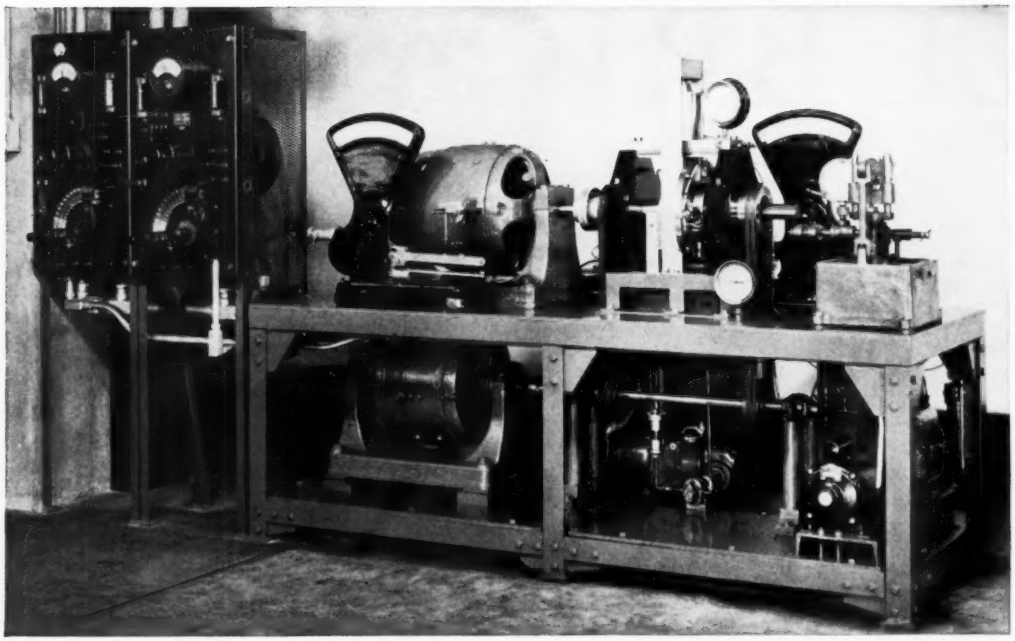


Figure 1 — Research type bearing test machine.

bution within the tapered oil film is shown in the surrounding polar diagram. Maximum pressure occurs near the point of minimum clearance between the shaft and bearing. It is to be noted that the location of this peak pressure is not directly under the shaft as the direction of its weight "W" would suggest but, with a clock-wise shaft rotation, is off-set in the direction of rotation to the lower left quadrant at 196 degrees.

The oil supply to this bearing, being in the unloaded half, was in a region of low pressure. Therefore, normal bearing supply pressures of 20 to 100 pounds per square inch would adequately feed such a bearing. If, however, the oil inlet hole could be moved around close to approximately 265 degrees where a region of negative pressure exists, the oil would be drawn into the bearing with no need for a force pressure feed. The placement of the inlet hole near 265 degrees would be mandatory for bearings fed by splash, drip or other non-pressurized means of application.

In certain types of bearing duty, particularly where surface speeds are high, where there are sudden changes in load direction and intensity, and where the available oil supply to the low pressure area is limited, the pressure may become so low in the negative pressure zone within the bearing clearance that the lubricating film ruptures. This momentarily produces unstable vapor pockets within the bearing clearance, an effect called *cavitation* which can destroy the bearing surface. As such damage usually appears on bearing areas away from the center of pressure, it frequently may be hard to account for and may be confused with a corrosive attack.

Figure 3 shows the loaded half of a Diesel connecting rod bearing which had experienced cavitation damage of a severity sufficient to remove metal from the bearing surface. The parting face of this shell was installed on a horizontal centerline and the maximum damage occurred slightly below the parting face in a location strikingly similar to the suction area of Figure 2 located five degrees below the horizontal centerline.

Grooving

In investigating many new bearing designs it seems that proper importance is not assigned to the location of the oil supply hole or groove. It is realized that the over-all design of the machine sometimes dictates their locations in positions that are less than optimum; however, when such a compromise must be made, greatly reduced load-carrying capacity frequently results.

In the past, it was the practice for the bearing surface to be laced with many inter-weaving or serpentine grooves which broke up the surface into many small areas; the more artistic the pattern, the better the bearing was believed to be. Bearing research has shown, however, that such grooving is useful only where lubrication is limited — where thin film conditions exist — as where the shaft motion is oscillating or subjected to frequent starts and stops.

Journal bearings running at more-or-less constant speeds or over narrow speed ranges (as is the more general case) usually can be lubricated adequately when the oil is supplied through a hole in the bearing or through a single axial or circumferential groove. As any form of grooving within the load-carrying zone of the bearing tends only to destroy the hydrodynamic film carrying the load, a single entry hole at the point of minimum pressure would be the ideal case. If a greater supply of oil is necessary — especially where additional cooling must be provided in bearings of small clearance ratio — an axial groove at the hole location should suffice; the groove aids in distributing the oil along the length of the bearing and may become increasingly more necessary in bearings of long length-to-diameter ratios.

The effect of a circumferential groove can have a decided influence on the load-carrying capacity of a journal bearing. A groove helps to distribute the oil; however, a groove through the loaded region serves to drain off the oil at that spot rather than to feed it in. As shown in Figure 4 which represents two conditions of the same total bearing load, a central groove causes higher peak pressures than the ungrooved bearing. At the same time, minimum film thickness is decreased. With an excessively wide groove the film thickness might be so reduced that abrasive solids in the oil cause undue wear. Furthermore, peak pressures may become so great as to physically damage the bearing surface.

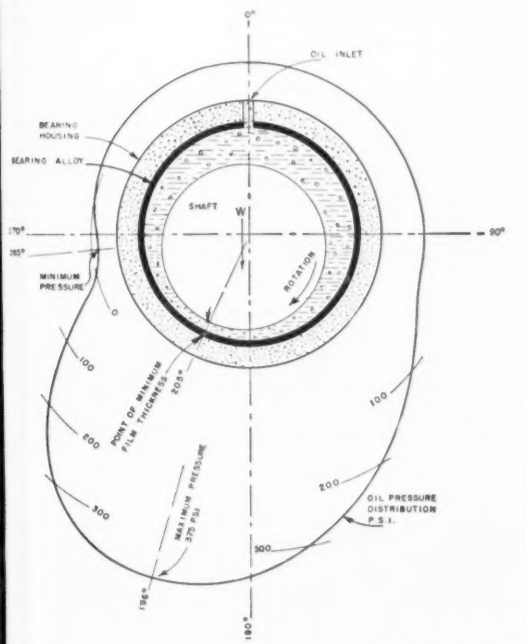


Figure 2 — Full floating journal bearing showing distribution of oil pressure under a vertical load.

Figure 3 — Connecting rod bearing showing cavitation damage near parting edge.

Proper grooving, however, has its advantages. As a circumferential groove permits a much greater rate of oil flow, the bearing may be safer in operation than its ungrooved equivalent. That is because the cooling is proportionately greater, the bearing metal is not so weakened by heat and the oil suffers a lesser reduction in viscosity.

If a reduced film thickness cannot be tolerated, a decrease in peak pressure would necessitate a reduction in load. One way of reducing the loss in load-carrying capacity resulting from a continuous groove is to have the groove continue for about three-quarters of the circumference and gradually thin out to a solid ungrooved surface through the general load-carrying area. If a bearing shell containing a partial circumferential groove is used in an application where the direction of load application varies over wide limits, difficulty might be experienced especially if the lubricant cannot be kept free of solid contaminants, which promote uneven shaft wear. At moments when the load direction causes the minimum film thickness to be toward the grooved portion, abrasive wear will be greater under the bearing surfaces that are adjacent to the groove. Although the wear rate may be slight, in time a pronounced ridge will be created on the journal and this ridge may then overstress the mating zone on the ungrooved bearing, reducing the fatigue life of the bearing shell in that area. Figure 5 shows an originally ungrooved bearing half that had been matched with a grooved half-shell in an internal combustion engine; a fatigue breakout through the center of the bearing is its attempt to readjust for a ridged shaft. Where bearing clearances are only a few thousandths of an inch, a ridge only a few tenths of that dimension may cause failure. Accordingly, when replacing bearings on such journals, they should be carefully resized and cylinderized.

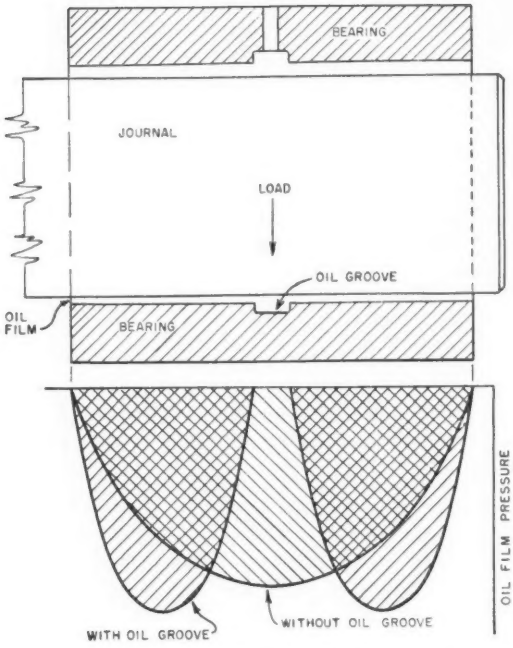


Figure 4 — A circumferential groove alters the pressure distribution within the oil film in the loaded region of a journal bearing.

THE BEARING SURFACE

A bearing may be considered as being only a physical support for a journal, but in serving that function it has to have certain attributes such as:

1. Minimum friction under conditions of marginal lubrication.
2. Sufficient plasticity to conform to irregular shafts and to embed solid contaminants too large to pass without rupturing the oil film.
3. Good thermal conductivity to carry away frictional heat from the rubbing surface.
4. Sufficient strength to carry the load and not to squeeze out or fracture by fatigue under cyclic stresses.
5. Be chemically inert to its environment.

Many of these attributes of necessity conflict with each other. No one bearing material meets all the requirements to best advantage. For example, low frictional properties, good plasticity, conformability and embedability are associated with the Babbitt alloys—the alloys containing major amounts of tin or lead. Good strength and fatigue characteristics may be found in the bronze, copper base, silver, and aluminum alloys; while the silver and aluminum alloys have the best thermal characteristics.

Through the years as the knowledge of bearing performance widened, design modifications have resulted in bearing operation of increased speeds and loads and greater life. Isaac Babbitt appears to be the first to take an active step to extend service life and improve load-carrying capacity of early bearings. In his U. S. Patent (1839) he suggested taking the brittania metal or pewter then in common use and *tinning* it into the bearing shell, thereby greatly improving the bond between the two metals.

As a typical alloy suitable for bearing usage, Babbitt mentioned a mixture composed of 50 parts tin (89.3%), 5 parts antimony (8.9%) and one part copper (1.8%); and the fact that it was an excellent but probably blind choice is attested by the record that today, over 100 years later, Babbitt bearings very closely resembling his composition are performing well in many large installations. The generic name of Babbitts, commonly known as *white metals* because of the color, now has been extended to include the lead-base alloys hardened with tin and antimony.

It is a credit to the modern bearing manufacturer that he can take advantage of the good points of several different alloys, blending them into a composite bearing that embodies the optimum characteristics of each metal. Such bearings, usually having a multi-layer construction, frequently are installed where the conditions of load, speed, temperatures, etc., are most severe.

Figure 6 shows the micro-structures of the fundamental layers of several multi-layer bearings. They are generally coated on a mild steel back or shell for good mechanical support, and may have a surface layer or overplate of lead, lead-tin, lead-indium or lead-tin-copper. Backings of cast iron or bronze also are used in certain installations, the latter frequently in marine service. The bearing alloy adjacent to the backing is chosen for its good strength and thermal properties: leaded bronze, copper lead, aluminum alloy or silver. The top layer, a thin film not more than 0.001-inch thick, usually is an electrodeposit. Frequent use is made of an overplate of an alloy of lead containing tin.

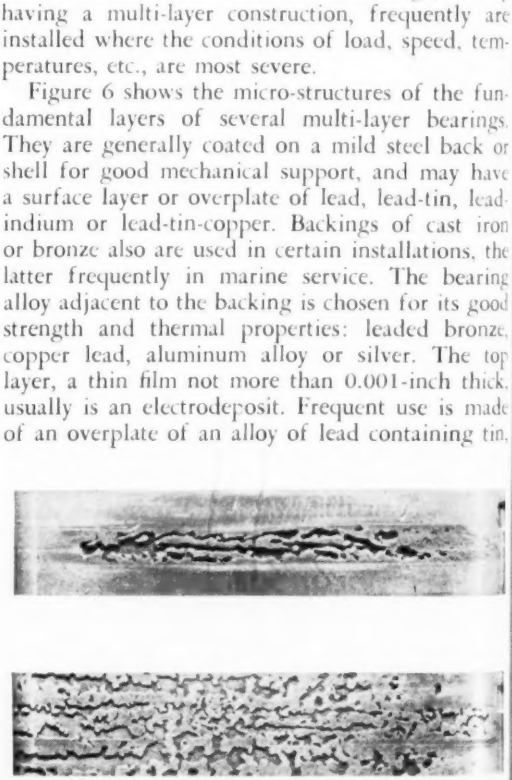
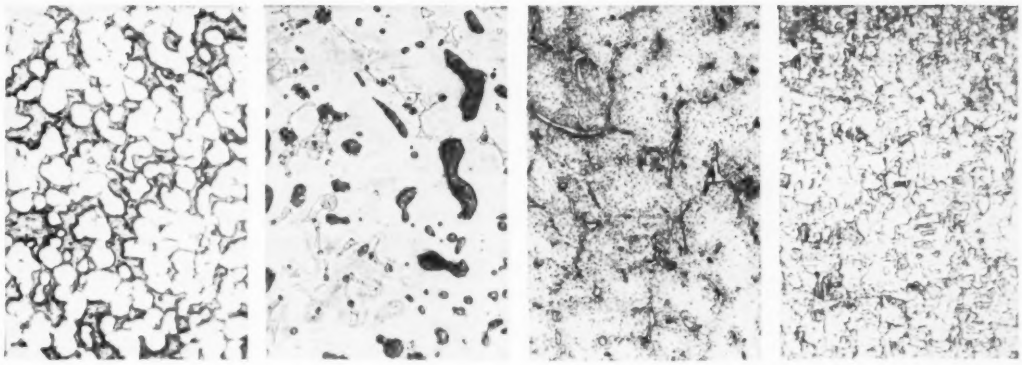


Figure 5 — Diesel main bearing showing fatigued central zone.



A — Copper-lead 100X. B — Leaded bronze 100X. C — Aluminum 500X. D — Silver 500X.

Figure 6 — Bearing microstructures.

and sometimes with a small amount of copper. The tin and copper improve mechanical strength and corrosion resistance. A small amount of indium diffused into the top layer will similarly improve the properties of the lead.

In planning a complex bearing material, the properties of each component metal must be known to insure mutual compatibility. For example, when coating a lead-tin overlay onto a copper-lead or bronze substrate, special provision must be made to keep the strata separated. That is because copper and tin are mutually very soluble in each other in the solid state. The tin of the overlay, therefore, has a tendency to diffuse into the copper alloy and, if uncontrolled, a lead-tin overlay with reduced tin content and changed physical and chemical properties will result.

Bearing temperatures within an operating engine frequently are sufficiently high to encourage this atomic movement; however, bearing manufacturers prevent the migration of tin to copper by placing a diffusion barrier between them. Usually an electrodeposit of nickel is used. This nickel "dam" is visible as a thin, wavy line in the photomicrograph of Figure 7.

Another situation where the unique attraction of copper for tin must be considered is where a high-tin Babbitt alloy is cast onto a bronze shell. When this is done, some of the tin from the Babbitt starts to dissolve and diffuse into the coppery bronze; the action gradually ceases as the casting cools. This atomic mixture at the Babbitt-bronze interface forms a new copper-tin alloy which is considerably harder and more brittle than the base bronze.

This self-created layer, white in color, can be seen in Figure 8 as the irregular crystals designated by an arrow. When present in a relatively wide band as shown, the copper-tin layer is more sensitive to fracture by overstressing and by fatigue; it may be the cause of a bond failure in service. More rapid cooling will reduce the tin diffusion to a safe limit.

BEARING OVERHAUL

Although the general subject of bearing overhaul is too extensive for a detailed treatment in this issue, there are a few points worthy of noting for those who rebabbitt their own bearings. With large industrial equipment the usual practice for the plant overhaul shop involves still-casting the Babbitt metal into a steel or cast iron shell. Even though the Babbitt ingots may be of a recognized analysis supplied by a reputable maker of bearing metals, poor adhesion often results. Therefore, to better anchor the Babbitt, certain operators favor grooving, undercutting or roughing of the steel backing; but such practice is not the best.

The initial hold may be secure in a grooved shell but operating loads set up stress concentrations at sharp ridges and corners. These stresses greatly reduce the fatigue life of the bearing alloy thereby causing early failure. Figure 9 shows a cross-section of a Babbitt bearing that had failed by fatigue. Notice how the cracks penetrate downward to the peaks of the rough-turned steel backing.

A better insight into the loading experienced by a bearing coated onto a roughened shell is illustrated in Figure 10, which represents the stress pattern of a model of an enlarged bearing cross-section obtained in a photo-elastic polariscope. The oval, dark bands gather about points of stress concentration and, in this model, it can be seen that they originate at each peak of the ridged steel backing. By this it is apparent that the peaks were stress-raisers which weakened the over-all bearing rather than improved it.

Good adhesion can be obtained on a smooth steel or cast iron backing by proper cleaning and tinning without grooving or otherwise deliberately

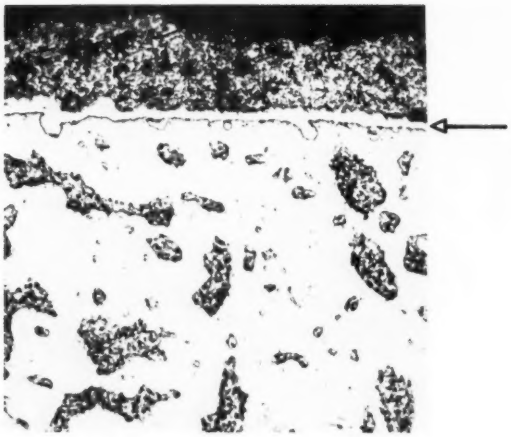


Figure 7 — Cross-section of heavy-duty Diesel bearing showing diffusion barrier (arrow) between copper-lead and overlay of lead-tin alloy. Magnification 500X.

roughening the surface. Cast iron shells, however, may require a preliminary degraphitization to provide an entirely metallic surface prior to tinning.

The structure of a poured bearing frequently can be improved by centrifugal casting. By this technique the mold or shell into which the bearing alloy is to be placed is rapidly spun about its longitudinal axis. The centrifugal force so created forces the molten alloy firmly against the shell; it reduces gas bubbles and generally enhances bearing properties. Caution, however, must be taken when centrifugally casting high-lead Babbitt because, if uncontrolled, a depreciated micro-structure may occur. If the chilling rate is too low or the spinning speed too high, pronounced segregation or separation of the micro-constituents may result.

Lead-base alloys are especially sensitive to this fault due to the greater differences in specific gravity between the several elements. The components which serve as hardeners — tin and antimony — and which also improve corrosion resistance of the lead form lighter-weight crystals which float to the top of the molten metal. The faster the rate of spin or the slower the chilling, the greater may be this separation. When the bearing face is subsequently machined much of this material is cut off, leaving the bulk of the bearing metal deficient in the hardening elements. This produces a bearing surface with inferior physical and chemical properties. Figure 11 shows a cross-section of such a segregated lead-base Babbitt bearing. The white line represents the location of the bearing surface if the shell were machined to leave a Babbitt thickness of 0.015 inch.

The plant overhaul shop should be aware that Babbitts of high *tin* composition also can be modified by their rate of cooling when cast. Slow cooling tends to produce larger crystals in the microstructure. The comparative micrographs of Figure 12 show this effect on an alloy closely duplicating I. Babbitt's original formulation. The fine structure was obtained by chill casting while the coarse structure was slow cooled, permitting a much longer time for the hexagonal copper-tin "snowflake" and needle-like crystals to form and grow. These crystals have much the same composition as the hard, white material at the bond layer in Figure 8.

ASPECTS OF LUBRICATION

The theoretical bearing designer is fortunate that the only lubricant variable he must consider is viscosity which is influenced by temperature and pressure conditions within the bearing. For the lubrication engineer, however, who has to recommend a commercial product for any specific bearing installation the lubricant question is far more complicated. He must take into account many apparently irrelevant machine and oil criteria, dependent upon the multiplicity of materials and conditions encountered by the lubricant. Only then, after carefully weighing all those factors, can he prescribe a lubricant compatible under all operating conditions. Among the more important factors for consideration are:

1. Contamination of the lubricant by the machine in question as well as by other contributing factors such as combustion soot, air-borne dust, moisture, chemical fumes, etc.
2. Life expectancy of the oil — how long it can endure between drains.
3. Secondary lubrication requirements; whether the bearing oil must lubricate other machine elements such as gears, pumps, cams, etc.
4. Chemical nature of the bearing surface and substrate.

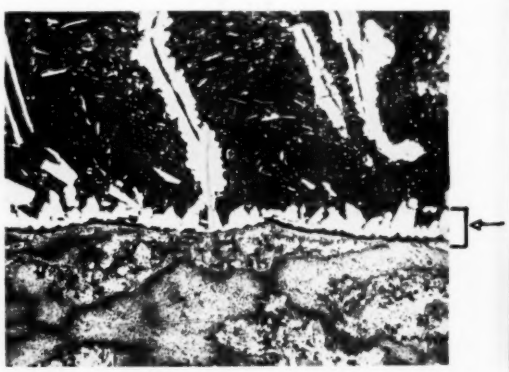


Figure 8 — Coarse copper-tin crystals at interface (arrow) between tin babbitt (above) and bronze (below). Magnification 500X.

Oil Contamination

Journal bearings ideally operating under hydrodynamic conditions do not dirty the oil; there is no metal wear and, theoretically, the oil should keep clean forever. In real life, however, the oil becomes contaminated. Every time the journal starts from rest, microscopic particles of bearing metal may be torn loose and picked up by the lubricant. Atmospheric dust can creep into the housing or loose material may become dislodged from the oil circulation system – all being picked up by the oil and, if not removed, they are potential wear-inducing agents.

If the bearing lubricant is contained within some larger, more complex machine, as for example an internal combustion engine, allowance must be made for contaminants peculiar to that operation. In fact, as is usually the case with internal combustion engines employing one lubricant for all parts, the combustion process so influences general lubrication that the lubricant requirements of the bearings within the engines often are secondary; combustion contamination may be the controlling influence.

Nevertheless, regardless of what contaminant may be present in a bearing lubricant, the engineer's problem reduces to:

1. He can select an oil that tends to keep solid contaminants in suspension so that they can be carried to some external filter or clarifying device for removal.
2. He can recommend an oil that allows ready separation for fast removal of solids.

Which course to follow depends upon the specific contaminants involved.

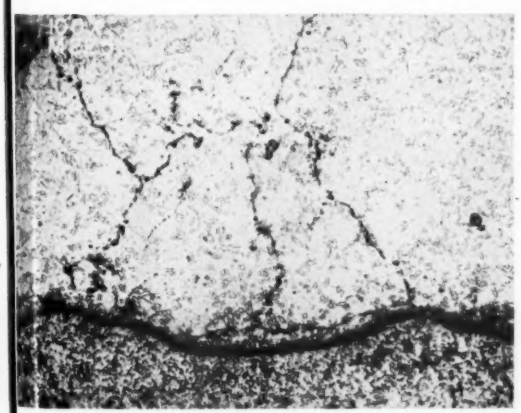


Figure 9 – Cross-section of a Babbitt bearing, failed by fatigue. Cracks are penetrating downward to the peaks of the rough-turned steel backing. Magnification 100X.

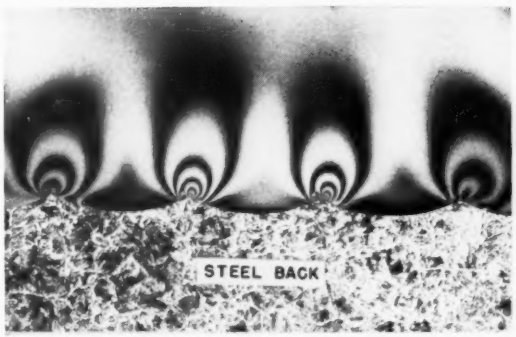


Figure 10 – Stress pattern in photo-elastic polariscope showing enlarged model of bearing alloy (upper) cast onto a roughened steel backing (lower). Note how each peak on the steel produces a stress concentration.

Turbine Oils

Steam condensate – water – is the principal contaminant to be guarded against in steam turbine lubricants. If allowed to accumulate in such applications, or in other large oil systems as in paper mills where water is perpetually present, it could build up into quite plastic emulsions which can plug oil lines. As a turbine oil circulating system may carry thousands of gallons, frequent drains and refills with fresh oil are not economically feasible. Therefore, water must be eliminated as fast as it collects. This requires a lubricant with rapid separation properties and negligible tendencies to form stable emulsions so that water, usually in the form of fine droplets, can be removed by filtration or centrifuging. Besides having good demulsibility, turbine oils must protect circulating systems from rust. Furthermore, for long life they must have good oxidation resistance so as not to thicken in service. Turbine bearings themselves generally are high-tin Babbitt and, consequently, are not affected by moisture.

Diesel Engines

In Diesel engines the demands upon a lubricant are so varied (and at times so contradictory) that its formulation is a very complicated matter. The requirements may be so complex, in fact, that larger engines with hundreds of gallons of oil in the system may find it advantageous to use two different lubricants, one for the lower crankcase including the crankshaft and rod bearings and a different oil for the piston, rings, and cylinders. Where such a separation can be made, journal lubrication is relatively simple, bearing temperatures are not high, and accordingly oxidation is slow. The only contamination to be guarded against would be from atmospheric dusts and moisture, bearing wear and a small amount of oxidized fuel and oil hydrocarbons dropping down from the cylinder lubrication. Filters or centrifuges can take care of these

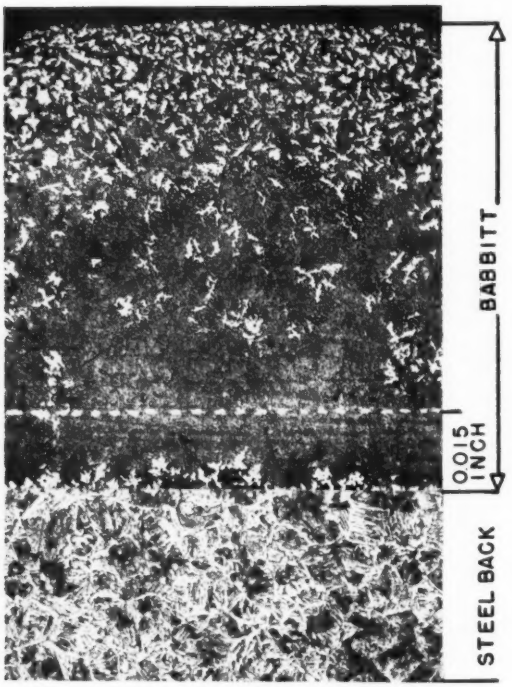


Figure 11 — SAE 15 high lead Babbitt centrifugally cast onto steel backing. Slow cooled, full thickness, as cast. Magnification 35X. Dash line represents final thickness when machined to 0.015 inch.

contaminants and maintain a crankcase oil in a clean condition for long periods. The limiting factors then would be thickening of the oil from slow oxidation.

In heavy-duty Diesel engines in which one common lubricant is used for all engine requirements the oil is faced with the problem of handling relatively large volumes of fuel soot along with the lesser amounts of the usual solid contaminants: wear metals, atmospheric dust, etc. With straight mineral oils in heavy duty Diesels, soot with other fuel oxidation products can rapidly dirty the engine, break rings, seize pistons, corrode bearings, etc. To protect against those many damaging influences, the contaminants must be removed from the oil or their effects cancelled out as rapidly as possible.

The larger solid contaminants such as particles of metallic wear, dust and dirt can be controlled easily by filtration; their rate of formation is low and normal filters can keep them cleaned out of the oil. Filters, however, soon are rendered inoperative by the relatively large volumes of agglomerates of fuel soot. As fuel soot in itself is too soft and too fine to damage bearings or other moving parts, there is no need to take that contaminant out of the oil if it can be prevented from forming large clumps or masses capable of plugging filters or other small passages.

The practical fix then is to so condition the soot that it will remain as fine discrete particles. This is done by using oils containing detergents or dispersants which have a peptizing action on the soot particles, preventing them from clumping together and, by so doing, keeping them in a fine state approaching a colloidal suspension. When finely dispersed in the oil, the soot then passes through the filter without being removed and without clogging the filter. Properly compounded detergent oils, therefore, leave the filters in an unplugged condition so that they can be effective in removing the solid particles and moisture that can do real harm to moving parts.

Dilution by fuel always must be considered with any lubricant for gasoline or Diesel engines. While the performance of engine bearings is not too critical with regard to specific oil viscosity, usually there is some minimum limit below which bearing operation will be unsafe. It will be appreciated that with engines in poor condition such a situation might be reached quite rapidly, when it is realized that the viscosity of a bearing lubricant will be roughly halved by the addition of ten per cent of liquid fuel.

Moisture

If the engine or machine contains journal bearings of high lead content (high lead Babbitts, alkali metal hardened lead or copper-lead) special provision may have to be taken to protect them against the effects of moisture. That is because the presence of small amounts of water in an oxidized crankcase oil may significantly increase its corrosiveness to lead. The increased attack caused by water is believed due to the increased ionization of the low-molecular-weight oxygenated hydrocarbons produced in the oil by oxidation, by compounds which have fair water solubilities and which have a solubilizing action on lead.

This effect is substantiated by laboratory tests shown in Table I, in which several different bearing metals were tested against an oxidized straight mineral oil and that same oil containing two per cent water. Note how, under either oil condition, the high-tin Babbitt is impervious to attack whereas the pure lead and the SAE 15 lead Babbitt* show significant weight losses. The alloying elements in the SAE 15 metal, however, greatly reduce the extent of the attack below that experienced by pure lead.

Moisture in additive oils may have additional complications. Some additives are water-sensitive, can break down to form stable emulsions or set

*SAE 15 Babbitt: Tin 1%, Antimony 15%, Copper 0.6 max. %, Arsenic 1%, Lead remainder (about 83%).

TABLE I

Corrosion Strip Weight Losses, Milligrams
200°F., 168 Hours With
Mineral Oil Oxidized To a Neut. No. of 2.0

Metal Strip Material	Oxidized Oil	Oxidized Oil + 2% H_2O
High Tin Babbitt (SAE 12)	0.2—0.2	0.0—0.0
High Lead Babbitt (SAE 15)	0.5—2.3	84—85
Pure Lead	370—375	384—392
Lead Babbitt, alkali hardened (contains calcium and tin)	134—149	286—299

up conditions conducive to bearing corrosion. At other times water — especially in the form of finely dispersed droplets — can negate some of the dispersive properties of the surface-active additive and thereby contribute to dirtier engines, clogged filters, etc.

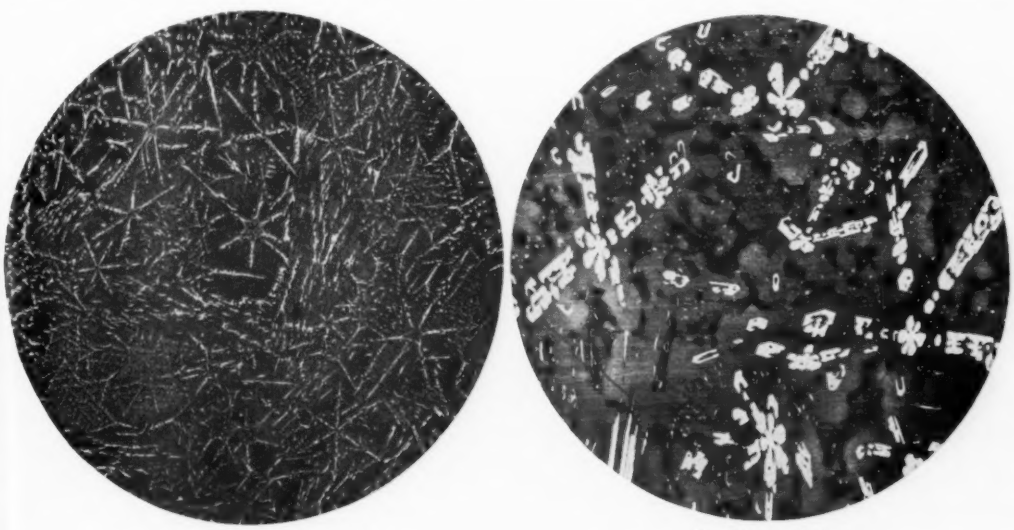
The obvious protection against these deleterious effects of moisture is to remove it as fast as it accumulates. This may require adequate filtration or centrifuging equipment, or operating the machines at temperature levels sufficiently high that the evaporation of water will be encouraged. Furthermore, the possibility of water contamination requires the lubrication engineer to choose carefully a lubricant with adequate water-separating properties; a lubricant which will contain no material susceptible to deterioration in the presence of water.

LIFE EXPECTANCY

How long a bearing lubricant is expected to give satisfactory performance often is an economic decision. An oil need be drained and replaced only when its characteristics have been so altered by use or contamination that it might be potentially dangerous to continue using it. Contamination by solids — already discussed — can be effectively controlled by filtration. Contamination by materials soluble in the oil are more difficult to handle. Sometimes they can be removed by heating or by adsorbent-type filters; however, they may accumulate to some arbitrary value based on experience, at which time replacement of the lubricant becomes necessary.

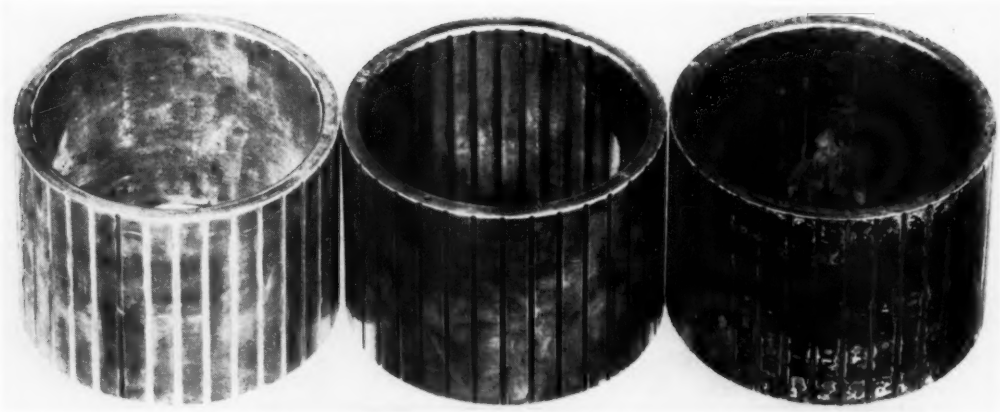
For large circulating systems oil renewal can be a costly matter. Accordingly, lubricants that can give maximum operating life are economically justified. Such oils usually contain oxidation stabilizers and corrosion inhibitors to fortify them against soluble contaminants.

Many other industrial applications exist where one common lubricant must lubricate journal bearings as well as other machine elements. In most instances the bearings could have been satisfied with a relatively simple mineral oil, but the demands imposed by the remainder of the machine often require an additive-type lubricant. As previously mentioned, a Diesel cylinder may require detergency; a steam turbine, rust and oxidation resistance; marine engines, water resistance. Gear sets, cams or vane pumps may require anti-wear and extreme-pressure properties. Because of these



A — Chill cast. B — Slow cooled.

Figure 12 — High tin Babbitt. Magnification 25X.



A B C

Figure 13 — Silver connection rod bushings showing effects of additive reactivity in heavy duty Diesel lubrication.

varied service requirements a bearing lubricant cannot be introduced without a full knowledge of the additive-to-metal reactivity of all the combinations involved.

For example, the detergent metal-organic compounds when used alone are pro-oxidants and are prone to instigate bearing corrosion. They require the supplemental use of an oxidation and corrosion inhibitor for complete protection and, for this service, sulfur and sulfur-phosphorus compounds have been found effective. One reason for their effectiveness is their ability to poison or de-catalyze new metal surfaces, thereby reducing catalytic effects upon oil oxidation. Closely associated with this effect is the tendency of these compounds to chemically react with metal surfaces such as silver, copper or steel, forming a metallic sulfide type of coating.

An illustration of how this metal additive reactivity must be considered in the development of heavy duty oils for special equipment can be shown in the results obtained with three experimental oils all of which satisfactorily lubricated journal bearings of the four types illustrated in Figure 6. It was necessary, however, that these oils also lubricate silver bushings operating at a considerably higher temperature. Figure 13 illustrates those bushings after 300 hours of operation on each oil. Oil A caused no deposit on the silver; its surface was bright and clean, but severely worn. Oil B formed a thin dark sulfide film which provided a durable wearing surface. Oil C caused an excessive chemical attack on the silver, in this case forming a much thicker silver sulfide film. With such a thick film, particles of silver sulfide tore out giving a corroded effect. Furthermore, the silver sulfide particles accumulated in the oil grooves which become clogged leading to oil starvation.

Other instances where metal-additive activity must be critically considered involve lubrication under severe loads or high temperatures, where the proper amount of surface activity must provide a chemical film or barrier at the critical spots. Too much activity, however, can be just as bad as not enough, as an uncontrolled corrosive attack may result as illustrated in Figure 14. This photograph shows the inner surfaces of several bronze valve guides lubricated by an additive oil which, although "over-active" to the bronze guides, proved perfectly satisfactory for silver journal bearing lubrication. An etched surface can be seen at the cool ends of all these guides indicating the chemical nature of the deterioration. While the corrosive action was due to a sulfur-type inhibitor, it should be stressed that not all types of sulfur-containing additives would cause such damage. The metal itself plays a big part in these reactions; guides of aluminum bronze are much more resistant to this type of attack than the tin bronze, while cast iron is unaffected.

POROUS BEARINGS

Solid journal bearings generally require a continuous oil supply to maintain a hydrodynamic film. There are circumstances, however, due to design limitations where it may not be convenient to provide a positive oil supply. In such medium or light duty applications, porous bronze bearings often prove advantageous because they carry their own oil. Such installations may include:

1. Bearings at remote points difficult to lubricate as a crane pulley or elevator guide rollers; or in business machines, electric timing devices, etc.
2. Bearings where oil drip from over-lubrication would be objectionable as in food machinery.

3. Bearings in domestic appliances and toys where lubrication often would be neglected as in electric fans, clocks, lawn mowers, toy trains, etc.
4. Bearings in small or inexpensive mechanisms where parts must rotate at a minimum of cost.

Porous bearings are made by pressing together powders, usually of copper and tin, and frequently including zinc, iron, lead and graphite. Sintering by heating them to a high temperature (but below the melting point) fuses that mix into a strong unit. By regulating the tightness of compaction, the volume of the voids within the bearing can be controlled over the usual range of 10 to 35 per cent by volume. That is, up to 35 per cent of the bearing volume in the form of inter-connected microscopic pores may be available for absorbing lubricating oil. These pores serve as a reservoir from which lubricant may be fed to the bearing surface as it is required. Replenishment of lubricant may be accomplished by bringing additional oil into contact with any part of the bearing – the bearing back for instance – where it will be readily absorbed and carried to the bearing surface.

From the standpoint of the petroleum marketer, the lubrication of porous bronze bearings is not as simple as it superficially appears. Not any oil of the proper viscosity will do, because the oil that is absorbed within a porous bushing is contained within microscopically small cavities. The oil actually is in contact with a relatively enormous metallic surface. That surface – usually bronze – is highly catalytic and accordingly a very oxidation-resistant oil is necessary.

GREASE LUBRICATION

When a lubricant is considered in the pure sense – to reduce friction or minimize wear – there is little classical difference between the performance of greases and oils. In either case a tapered hydrodynamic film is ideally created to carry the load. Laboratory tests on research bearing testers such as the one shown in Figure 1, show that there is a pressure distribution through a grease-lubricated bearing similar in form and position to that in an oil-fed bearing, with the exception that the pressure peaks may be flatter and side leakage somewhat reduced.

There are certain instances where grease lubrication may prove advantageous. These include:

1. Applications where relubrication may not be convenient as in bearings hard to reach. The reduced side-leakage of greased bearings makes a small amount of grease stay put longer.

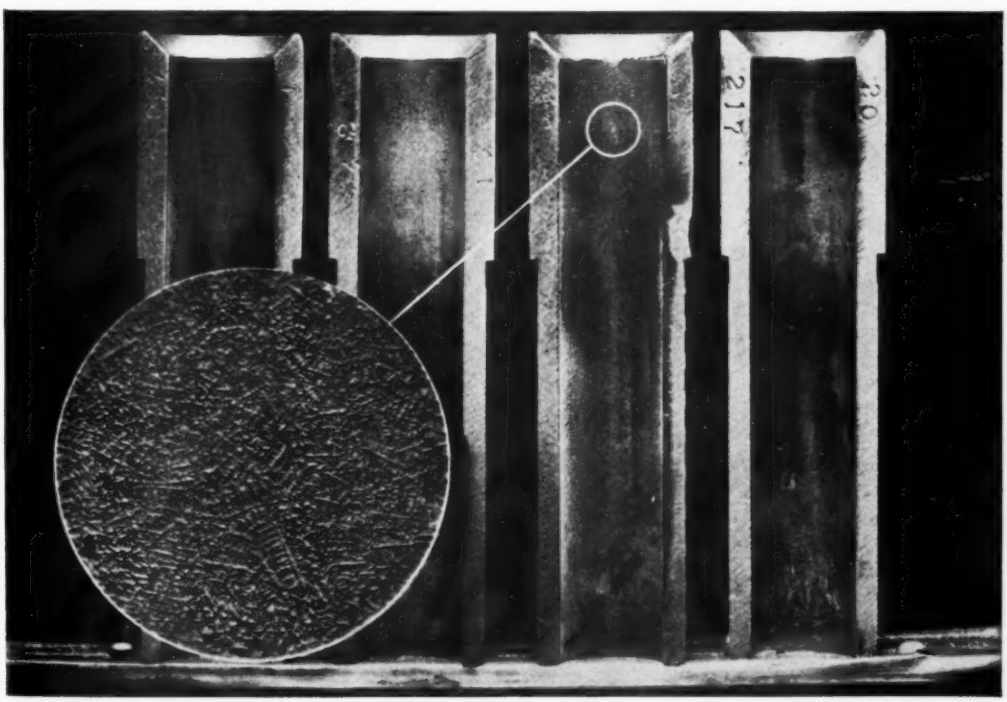


Figure 14 – Bronze valve guides showing surface damage when lubricated with over-active additive-type oil.

2. Bearings operating in dusty or abrasive environments where a collar of grease worked out of the side of a bearing quite effectively seals it to the ready entrance of hard, film-destroying contaminants. This ability to form a seal is especially advantageous on rolling friction bearings which are extremely sensitive to damage by hard solids.
3. Bearings where dripping may be a problem as in food processing equipment, textiles, and paper mill operations.
4. Correct choice of a grease with regard to its water repellent properties can also make it an effective seal against moisture.
5. Operations where the cooling effect of circulating oil is not necessary.
6. Operations where the bearing encounters high shock loads, high temperatures or extreme pressures, with frequent stops and reversals as in certain rolling mill duties. Greases are advantageous in such instances because of their greater resistance to squeezing out without rupturing the lubricating film. Greases also can have incorporated in them additional compounds which increase load-carrying or anti-seizure properties.

SUMMARY

Keeping all the aforementioned points in mind, the lubrication engineer is then in a position to recommend a specific product. He can do that with assurance because of his intimate knowledge of all phases of the operation as well as of his knowledge of the complete behavior of the lubricant involved. He can recommend the oil (or grease) that will be best for the equipment if he is not limited by arbitrary specifications set up by the consumer.

In the past it has been the practice of the purchaser of industrial lubricants to judge quality by how the product met certain analytical standards, physical or chemical tests as, for example, color, neutralization number or ash. For straight mineral oils such comparisons gave a fair measure of quality, but with the increasing use of additive-containing oils, the "quality" picture is becoming harder to judge. For example, a bright, clear, golden or light color was once regarded as a mark of refining quality, but today the best oil for a certain application may be quite dark, or it may have a very unusual color. That is because many additives are inherently dark or almost black whereas others are bright colored.

An ash specification can be similarly restrictive. In the pioneer days of petroleum technology ash content was a measure of abrasive mineral matter, earths, clays, etc., not completely removed due to refining limitations. Today, however, refining techniques have so improved that it is easy to produce a well-refined oil essentially ashless in composition. In a new oil, "ash" now serves only to reveal the presence of the metallic component of organic additives which are completely soluble in the oil, which are not abrasive in themselves and which, by virtue of their film-forming properties, actually may have the ability to reduce wear.

Neutralization number also is losing much of its original significance. For many years it has been a time-honored means for measuring the concentration of acidic materials present in an oil which could contribute to corrosion; but that importance is diminishing as, by modern refining techniques, it is easy to produce lubricating stocks that are essentially neutral – that is, having neutralization numbers equivalent to less than 0.05 milligram of potassium hydroxide per gram of oil.

However, when additive oils entered the picture the entire concept of neutralization number was upset. Because of the way in which it is determined, certain additives react with the test reagents so that a fictitious neutralization number is indicated even though the additive either alone or in combination in the base oil produces no corresponding corrosive action on metal. It therefore behooves the purchaser to look to the reputable petroleum marketer for assurance that the oil contains no components deleterious to the equipment, regardless of the finite values of any physical or chemical test determinations.

It is recognized more and more that metal-additive reactivity has certain definite functional aspects which contribute to the excellent performance properties of additive oils. For these oils to perform to their maximum usefulness, however, their usage should be limited only by the equipment itself.

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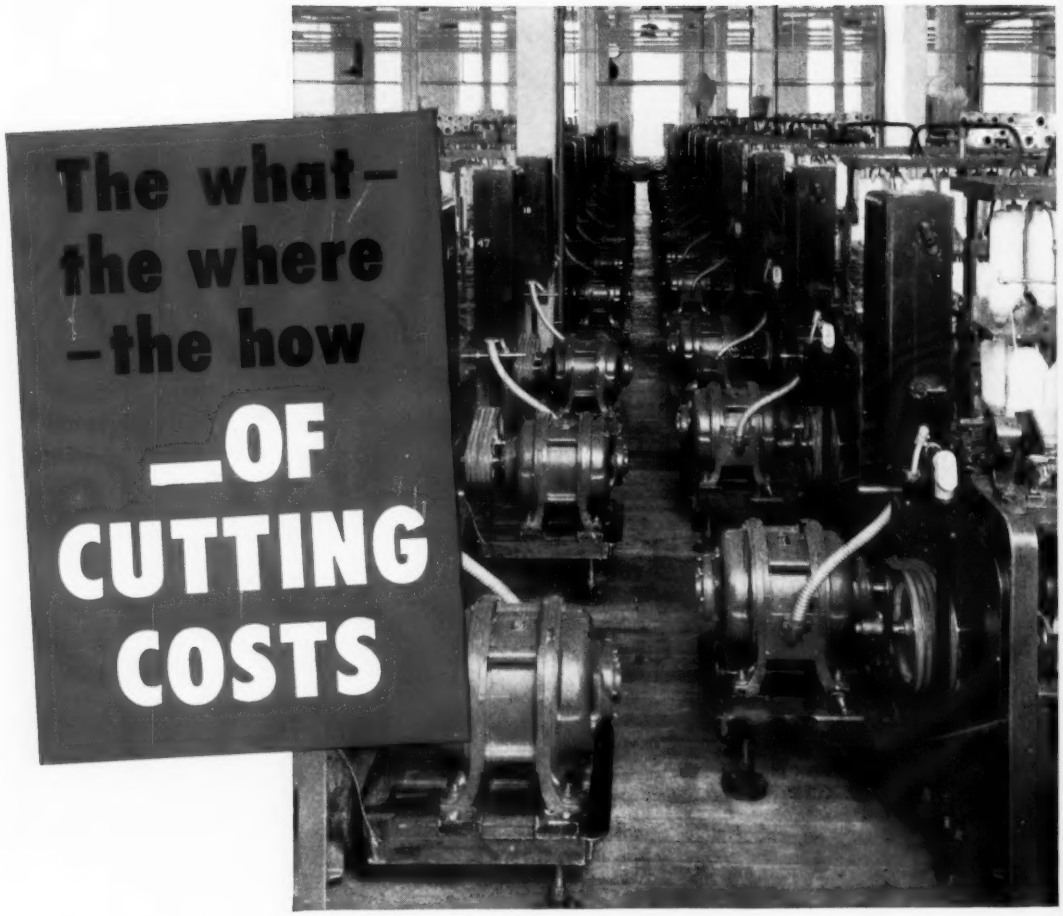

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